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Abstract

As part of the Space Laser Energy Program proposed by the National Aeronautics and Space Administration (NASA), the Jet Propulsion Laboratory (JPL) has recently completed a study on the Beam Transmission Optical System (BTOS) which is the system employed to deliver laser energy from Earth to space targets. The purpose of this study was to 1) identify the driving environmental and functional requirements, 2) develop a conceptual design, and 3) perform static, thermal distortion, and modal analyses to verify that these requirements are met. The study also identified major areas of concern which should be investigated further,

trajectory path. Finally, SELENE will provide continuous steady-state energy for operation of a lunar base.³

The free electron laser development is a major portion of the SELENE project which is being managed by Marshall Space Flight Center. The power beam is characterized as a Gaussian distribution which is cored in the center to prevent reflections from returning to the source and is truncated at its edge. The ten megawatt beam expands from the laser port through natural expansion in a vacuum tube over one kilometer in length until the beam becomes one meter in diameter and has a Strehl ratio of 0.9 or greater.⁴

1. Introduction

The development of the Beam Transmission Optical System (BTOS) is a portion of a larger project entitled Space Laser Electric NEergy (SELENE). The SELENE project utilizes a high energy, free electron laser to transfer energy from the ground to orbiting spacecraft or other space targets such as a lunar base.¹ BTOS is the system that delivers the beam energy from the laser to the target,

The primary mission objective of SELENE is to provide energy for operation of geosynchronous satellites including steady-state power for operations, periodic low power for station keeping, periodic high power during eclipses, and high power for transfer orbit apogee burn.² SELENE will also provide energy for operation at middle and high earth orbits (MEO) of 3000+ kilometers. Another possible usage for SELENE will be to provide energy to a laser-augmented solar-electric orbit transfer vehicle wherein a low earth orbit (LEO) vehicle transfers to geosynchronous orbit (GEO) through a spiral

Functional design requirements for BTOS are drawn from the most taxing case from each intended mission. The aperture size was determined from power requirements of the lunar mission and is currently set at 12 meters. The optical design is an on-axis Cassegrain system with a baseline f-number of 1.25.⁴ Slew rates and accelerations will be set by MEO missions. To provide for the necessary power requirements at the target, which include an overall Strehl ratio greater than 0.5, it is necessary for the beam path to correct for atmospheric disturbances.^{5,6} Atmospheric disturbances include natural wind driven thermal gradients and thermal blooming effects caused by the beam itself.⁷

Atmospheric correction for the BTOS project is accomplished through the usage of an active, segmented primary mirror. The diagram in Figure 1 illustrates how the system works.⁴ For the current site location (White Sands, New Mexico) the r_o is estimated to be three centimeters (3 cm). The initial design for the primary mirror requires the usage of over 150,000 hexagonal, 3 cm flat-to-flat mirror segments, each of which is capable of being commanded in tip, tilt, and piston by utilizing three voice coil actuators.^{8,9} These commands will be made by a control system with a 300 hertz bandwidth. Edge sensors and the wavefront sensor

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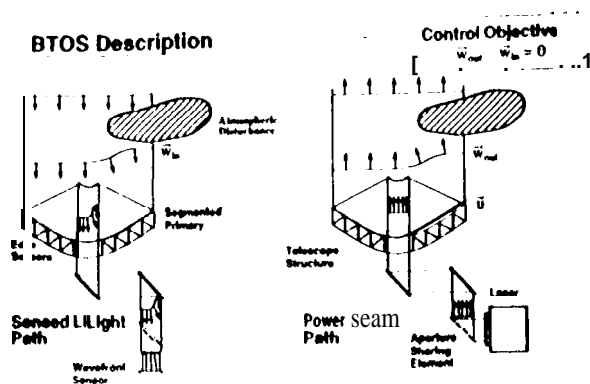


Figure 1. Atmospheric correction schematic

are used to reconstruct the output wavefront match edges at an update rate of up to 3000 hertz. The commands are generated by the wavefront control computer using information from the wavefront sensor.”

In addition to corrections for the atmospheric disturbances, the BTOS system must also correct for environmental effects on the telescope structure such as thermal distortion, gravity effects, and wind loading. The gravity effects come about due to the possibility of pointing in any direction from zenith to 20 degrees above horizon. By making a stiff primary mirror support structure, the deflections can be minimized. Rigid body tip and tilt (the first two polynomials of the Zernike coefficients) can be corrected by the pointing control system and/or by a one meter diameter, actuated tip/tilt mirror. This mirror is currently designated as the elevation mirror. Current plans for the design incorporate the use of 156 hexagonal cluster panels each supporting about 1000 mirror segments. These cluster panels may also be actuated through a separate metrology system to correct for static distortions,

This paper describes the efforts leading to a baseline design of the BTOS structure and identifies areas of concern requiring further effort. Many considerations must be blended together to form a workable design. The distortion of the primary mirror was the overwhelming concern, and

minimizing the distortion became a major design driver for the BTOS structural system. Distortion of the primary mirror is caused by both thermal gradients and gravity effects. To reduce temperature variations across the mirror surface, incorporation of thermal control into the design will be necessary. To minimize distortion caused by gravity, efforts were made to achieve a lightweight, stiff structural design. Another important consideration for the BTOS support structure was dynamic interaction with the 150,000 actuated mirror segments and other actuated optical elements. Modal behavior of the structure was thoroughly characterized and it was concluded that the need for passive damping augmentation should be addressed through a series of dynamic tests.

11. Scope of Study Effort

SELENE is a project involving many disciplines and requiring the cooperative efforts of many organizations. Marshall Space Flight Center delegated the responsibility for developing the overall BTOS system to JPL. Responsibility for developing the individual mirror segments remained with Marshall Space Flight Center and therefore “discussions related to segment design will not be included in this paper.

The BTOS system does not work properly without the presence of a dome structure.” The design of the dome structure will not be addressed in this paper, however, certain assumptions regarding the dome must be made in order for design of the telescope system to begin. It is assumed that the dome 1) would have an aperture that was capable of following the motion of the laser beam as the telescope system was articulated, 2) was fully enclosed with a thin, transparent, low conductive window in order to entrap a dry nitrogen atmosphere, 3) was capable of resisting all wind loading conditions for both operating and non-operating periods, 4) was able to resist inclement weather such as rain and snow, and 5) contained an atmospheric control system which would effectively circulate the dry nitrogen atmosphere and thereby tend to reduce the thermal blooming effects within the beam path,

Figure 2 shows a schematic of the BTOS telescope conceptual design. The hardware configuration of this design was developed based on the functional requirements and concerns listed above.^{11,12} The telescope consists of three basic components which are the mount structure, the tipping structure, and the optical elements.

Mount Structure

The mount structure (**alidade**) is a standard azimuth-elevation design. It is composed of welded, heavy steel pipes. Gravity forces are reacted through four wheels ("**trucks**") and lateral forces are reacted through the central **pintle** bearing. The **pintle** bearing is larger than one meter in diameter to allow the laser beam to pass through. A smooth, fast azimuthal motion is accomplished by utilizing the central bearing and a large diameter circular track with two drives. Elevation rotations are accomplished through a set of two elevation bearings and a friction wheel drive.

Tipping Structure

The tipping structure consists of the primary mirror support truss, the secondary mirror support structure, the tilt beam, and the counterweight/drive wheel structure.

The design of the primary mirror support truss was the major focus of the conceptual phase study. The baseline design assumes that deflections of the support truss can be compensated by a separate metrology system. Because the system will have capability to correct for distorted shapes caused by gravity sag or thermal distortion, a deflection limitation was never set for the primary mirror support truss. **However**, to minimize the stroke requirements of the separate metrology system, a reasonably stiff primary mirror support structure was emphasized. The primary mirror support truss is composed of over 1300 graphite-epoxy tubes bonded to low coefficient of thermal expansion (**CTE**) steel alloy end fittings. The tube **assemblies** are pinned to steel cluster fittings. Earlier versions of the design utilized welded aluminum tubes, but that design led to large radial excursions along the surface of the segmented mirror caused by the thermal

environment. The primary mirror support truss interfaces to the tilt beam structure at four locations which incorporate a radial compliance design. **This** compliance allows the tilt beam to be made from low cost steel which has a **CTE** different from that of graphite-epoxy. The primary mirror support truss also provides direct support for the elevation fold mirror in order to reduce the relative deflection between the elevation mirror and the primary mirror vertex,

The secondary support structure is composed of six graphite-epoxy box beams which are relatively narrow to minimize shadowing effects on the **primary** mirror. The configuration is essentially three bipeds which **attach** to the six corners of the primary mirror at one end and attach to a stiff, box-like, secondary mirror support structure at the other end. The structural beams are tapered in a manner that minimizes the shadow effects towards the center of the laser beam wherein the energy levels are more concentrated. The underside of the graphite-epoxy beams will **be** covered with a specular finish to reduce the heating effects caused by the laser beam. If necessary, the secondary mirror backing structure will **be** momentum **compensated** to minimize jitter caused by motions of the actively controlled

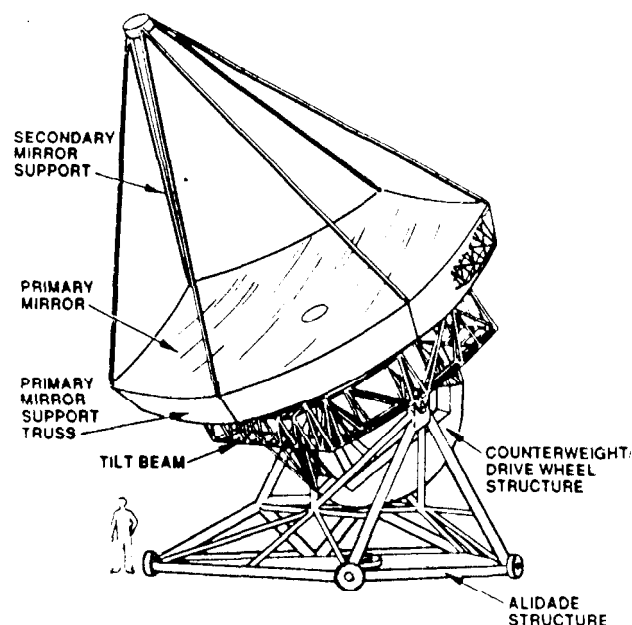


Figure 2. BTOS telescope configuration

secondary mirror.

The tilt beam provides structure which transfers loads from the primary mirror at four locations to the two elevation bearings-on the **alidade** structure. The tilt beam is an assembly of welded steel tubes.

The counterweight/drive wheel structure is composed of welded steel tubes and plates with heavy masses attached to the structure for counterbalancing the tipping structure. The 132 inch radius elevation drive track is attached to this structure.

Optical Elements

The schematic drawing in **Figure 3** shows the major optical elements in the **BTOS** system.⁹ Each element must be held in place by stiff structures. The major elements of the optical design include the azimuthal beam splitter, the elevation mirror, the secondary mirror, and the set of corrective optics to reduce the beam size for a smaller diameter (5 cm) wavefront sensor.¹³

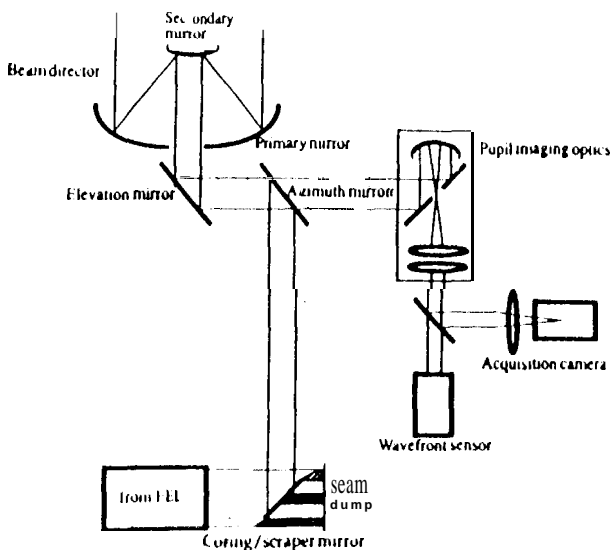


Figure 3. Optical element schematic

The **azimuthal** beam splitter is in the shape of an ellipse with a minor diameter of 1 meter and a major diameter of 1.5 meters. A **suggested** design calls for a fused silica plane with optical coatings which have the ability to reflect the outgoing laser beam and remain transparent to the incoming

beacon signal, however, the method of beam separation has not been selected. The azimuthal mirror (estimated at 40 kg) would be supported by a stiff 1.2 meter diameter steel pipe attached to the **pintle** bearing so that the azimuthal mirror moves with the **alidade** structure.

The elevation mirror is a solid elliptical mirror with the same shape as the azimuthal mirror. It is water-cooled to maintain dimensional stability under extreme heating conditions and could be made from molybdenum. It is mounted to the primary mirror support truss so that relative motions between the elevation mirror and the primary mirror vertex will always be small. It is designated as the tip/tilt mirror. The large mass associated with this mirror operating at high frequencies demands that a momentum compensation device must accompany this mirror.

The secondary mirror is a precise, water-cooled, one meter diameter parabolic mirror that is actuated in six degrees of freedom at frequency of actuation slower than two cycles per second. This mirror would be commanded by the separate metrology system.

The corrective optics "package" is mounted to the tilt beam next to the azimuthal mirror. There are no major **thermal** problems associated with these optics because they are not in the path of the laser beam. This group of optics will have a separate structural support to help maintain the tolerances between the individual optic elements. It has not been determined whether or not this "package" would need to be articulated for small corrections.

The 150,000 + individual 3 cm (flat-to-flat) hexagonal mirror segment assemblies are the responsibility of the Marshall Space Flight Center. These complex devices include the highly precise silicon carbide mirror, three voice coil actuators, three edge sensors, and a electronics. These assemblies will be mounted to 156 hexagonally shaped clusters that are either aluminum **facesheet/aluminum** core or graphite-epoxy **facesheet/aluminum** core composite panels. For the configuration **analyzed**, the mass of the active primary mirror surface was estimated at 48.82 kg/m² (10.0 lb/ft²). This estimate includes a 10.16 cm (4 inch) thick aluminum honeycomb composite panel

along with the hexagonal mirror assemblies. The cluster panels will be mounted to the primary mirror support truss at three locations in a kinematic fashion such that deformations of the truss structure will not distort the **cluster panels**. A separate metrology system will be used which will monitor low frequency (less than one cycle per second) disturbances such as gravitational and thermal effects. This system will send corrective commands to the cluster panel actuators which support the 156 cluster panels. The mass of this primary mirror is quite low compared to other monolithic mirror designs, which led to an incredibly low overall mass and inertia.¹⁴

Mass Properties

A preliminary estimate of the baseline conceptual design mass properties is given below.

<u>Item</u>	<u>Mass (kg)</u>
Alidade structure	11448
Base structure	(10889)
Azimuthal mirror + support	(559)
Tipping structure	27970
Primary mirror surface	(5378)
Primary mirror truss	(1337)
Secondary mirror + support	(326)
Counterweight/drive structure	(15565)
Tilt Beam	(4146)
<u>Optics + support</u>	<u>(1218)</u>
Total	39418 kg

An initial effort to size the elevation and alidade motors was performed with an earlier design that utilized an f-1.5 configuration with a heavier aluminum primary mirror truss. These values are conservative.

The alidade motor must slew the entire telescope and alidade structure in an azimuthal motion. The greatest demand is placed on the motor when the tipping structure is pointed lowest to the horizon. The total mass is estimated to be 44147 kg. The matrix below ($\text{kg}\cdot\text{m}^2$) was used to size the alidade motor:

$$I = \begin{bmatrix} 1.68 \times 10^6 & -5.56 \times 10^6 & 2.72 \times 10^5 \\ 5.56 \times 10^6 & 1.42 \times 10^6 & -1.12 \times 10^5 \\ 2.72 \times 10^5 & -1.12 \times 10^5 & 7.84 \times 10^5 \end{bmatrix}$$

The elevation motor must slew the entire tipping structure from 19.5 degrees above horizon to zenith. The mass of the tipping structure was estimated at 36632 kg. The matrix below ($\text{kg}\cdot\text{m}^2$) was used to size the elevation motor:

$$I = \begin{bmatrix} 6.41 \times 10^5 & 2.43 \times 10^{-1} & -8.05 \times 10^{-1} \\ 2.43 \times 10^{-1} & 6.29 \times 10^5 & 6.73 \times 10^1 \\ -8.05 \times 10^{-1} & 6.73 \times 10^1 & 2.73 \times 10^5 \end{bmatrix}$$

IV. Structural Desirer Requirements

Requirements placed on the telescope structure can be categorized into two major areas: environmental and functional,

Environmental design requirements include gravitational effects, wind loads, earthquake loads, and loads caused by thermal gradients. The gravitational deflections of the secondary mirror will cause the mirror to move axially and laterally depending on the elevation angle of the tipping structure. Wind loads were not analyzed based on the assumption that the dome structure would fully enclose and protect the telescope structure. Earthquake loads must be accounted for, but were not analyzed in this study effort. Base gravitational accelerations of 0.33 G's would normally be used, but the amplification factor for such a tall structure could create much higher loads. It is anticipated that strength will not be a problem for the telescope structure. Thermal gradients will typically not cause a strength problem, but could easily cause relatively large distortions of the primary mirror surface. Due to the immaturity of the design, a proper thermal analysis was not conducted, therefore no distortions caused by thermal gradients (or bulk temperature changes) were calculated. The high temperature environment caused by the laser beam will require an intensive study of thermal management and its relationship to active control of the optical elements.

Functional design requirements are imposed by various systems within the BTOS project. Because BTOS is an actively controlled system the major functional requirement focused on dynamic characteristics which could potentially couple with the active control of the individual primary mirror segments. The operating frequencies for the individual mirror segments range from 0 hertz

(static) to a maximum of 3000 hertz. Another functional requirement is relevant to distortion of the primary mirror surface. The primary mirror surface must maintain its original parabolic shape within the stroke capability of the individual mirror actuators, which is approximately one (1) mm,

A list of recommended frequency requirements and their avoidance consideration are given below, Most of these requirements were self imposed by the BTOS design team during the conceptual phase of the project.

<u>Item</u>	<u>Minimum Frequency (Hz)</u>
Secondary mirror support	10 Hz (wind buffeting)
Primary mirror support	15 Hz (wind & 2ndry supt.)
Cluster panels	100 Hz (global segment motion)
Mirror segments	15000 + Hz (high end vibration of actuators)
Elevation mirror with momentum compensation	500 + Hz (tip/tilt correction bandwidth of 300 Hz)

V. Structural Modelling

To address concerns related to strength and stiffness, a series of detailed MSC/NASTRAN finite element model was generated. Some modelling focused on preliminary designs for the individual mirror segments and the cluster support panels. These models were used to verify that segments and cluster panels could be made that met the functional stiffness requirements. A detailed BTOS model was used to generate distortion data for the effects of gravity, and to evaluate the overall strength and modal characteristics. The model employed 563 grid points and 2258 elements (1912 bars and 346 plates). Each element was assigned section properties and material properties. A total of 30 property descriptions were employed. Sizes ranged from 1.50 diameter, .113" thick graphite-epoxy tubes in the primary mirror support truss to 12.75" diameter, .50 thick A36 steel pipes in the alidade structure.

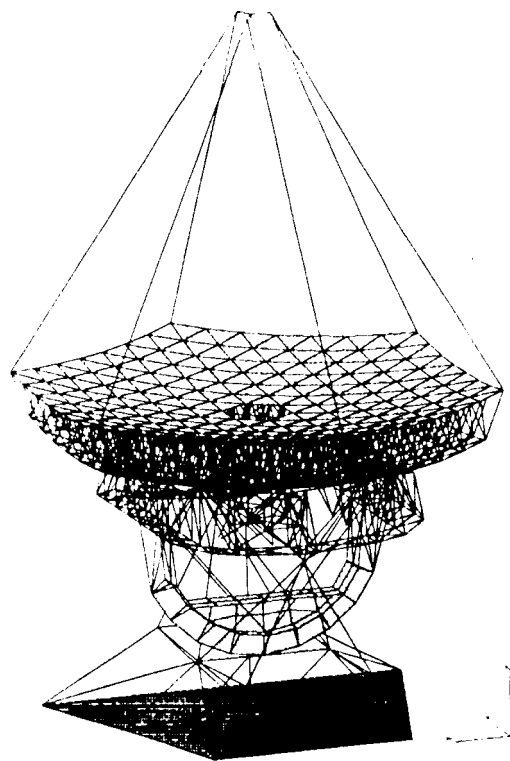


Figure 4. Plot of MSC/NASTRAN model

Counterweight masses totaling 15,565 kilograms were used to balance the tipping structure about its elevation bearings, Figure 4 shows a plot of the model.

VI. Modal Characteristic

To assess the dynamic characteristics of the entire BTOS structure, an analytical modal survey was performed, The results are shown below.

<u>Mode</u>	<u>Freq.</u>	<u>Description</u>
1	0.0 Hz	Tipping structure rotation θ X
2	4.3 Hz	Counterweight \pm X; Prim. mir. θ Y
3	6.8 Hz	2ndry mir. \pm X; Prim. mir. θ Y
4	7.1 Hz	2ndry mirror structure \pm Y
5	8.2 Hz	2ndry mirror structure \pm Y
6	8.3 Hz	Tilt beam bending \pm Z
7	8.8 Hz	PM twist (potato chip shape)
8	10.3 Hz	PM and tilt beam potato chip
9	12.4 Hz	Azimuthal mirror support \pm Y

Other models were generated of certain components. The secondary mirror support structure

was modelled in detail and produced six identical modes at 4.16 hertz and six identical modes at 5.39 hertz. These corresponded to lateral bending modes of the individual support beams in the weak and strong axes respectively. The first global mode occurred at 16.47 hertz which was a lateral motion of the secondary mirror relative to the primary mirror. The goal of 10 hertz was exceeded for this simplified model, however, when this model was coupled with the overall BTOS model the frequency fell below 10 hertz. A design modification will be required to correct this problem. Preliminary models of the cluster support panel showed that 100 hertz frequencies were achievable with simple honeycomb panel structures. Preliminary models of a silicon carbide lightweighted mirror segment showed fundamental frequencies greater than 30000 hertz.

VII. Gravity load Analysis

Strength Analysis

The largest stress in the BTOS structure for an earth gravity load (7. direction) is 7997 psi and occurs in one of the tilt beam pipes.¹² For a potential earthquake side load of 0.33 G's in the X direction, the maximum stress of 4935 psi occurs in the counterweight structure. When the tipping structure is pointed at 19.5° above the horizon, the corresponding 0.94 G's in the Y direction causes a maximum stress of 9131 psi in one of the tilt beam pipes. These stresses are very small compared to typical allowables for the chosen materials, and as such are not a problem.

Deflections of the Primary Mirror Support Structure

The BTOS segmented control system and cluster metrology system will provide displacement correction capability for the segmented primary mirror to perform its atmospheric correction function. Because of this feature, the support structure is not required to have as much stiffness as a design which incorporates a monolithic mirror or fails to incorporate adaptive optics into the system.¹⁴ A structural analysis was performed on the preliminary BTOS configuration to assess the stroke requirements for the cluster metrology system. The results indicated that maintaining a near perfect primary mirror during all elevation angle positions

placed too much of a burden on the segment actuators which led to the current baseline decision to utilize the separate metrology system described earlier. The two tables below indicate the maximum displacements of the primary mirror support structure when the tipping structure is pointed at zenith. Table 1 provides displacements relative to a point on the ground. Table 2 provides displacements relative to the primary mirror vertex such that rigid body displacements and rotations have been subtracted. Additional analysis was performed for the lowest (19.5° above the horizon) configuration in which the displacements relative to the primary mirror vertex were approximately double in all directions. Figure 5 shows a plot of the

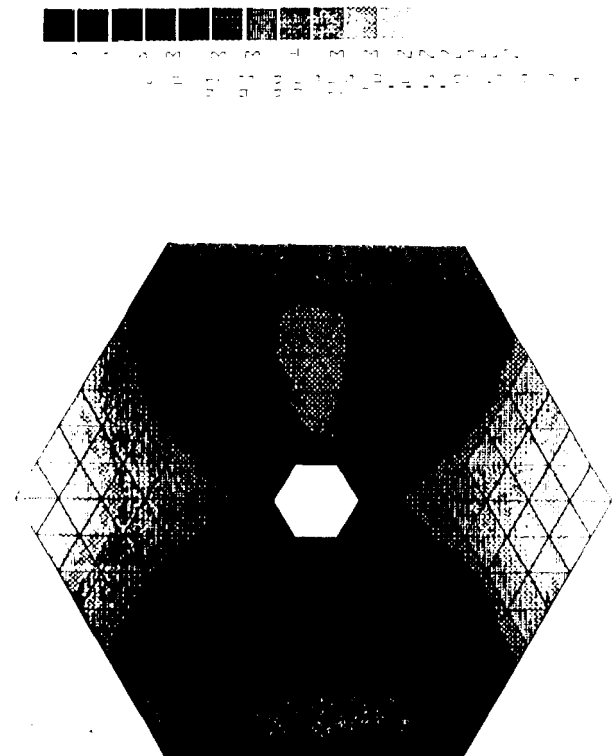


Figure 5, Z axis displacements relative to the primary mirror due to a gravity load (inches)

compensated Z axis (normal to surface) displacement. Note that the displaced surface is a maximum at the four tilt beam attach points.

Table 1. Zenith pointing configuration; displacements relative to ground				
Item	NASTRAN Grid Number	Maximum Displacements (inches)		
Primary mirror support truss	12018	<u>0.0108</u>	0.0010	-0.0982
	10215	0.0030	-0.0130	-0.1132
	12044	0.0025	-0.0051	<u>-0.1391</u>
Elevation mirror	5001	0.0031	0.0001	-0.0490
Azimuthal mirror	5502	0.0057	0.0000	-0.0223
Focusing mirror	5004	0.0183	0.0009	-0.1200
Secondary mirror	9001	0.0086	-0.0798	-0.2233

Table 2. Zenith pointing configuration; displacements relative to primary mirror vertex ^[1]				
Item	NASTRAN Grid Number	Maximum Displacements (inches)		
		X	Y	Z
Primary mirror support truss	12018	<u>0.0065</u>	0.0117	0.0088
	12008	0.0018	<u>0.0163</u>	-0.0004
	12044	-0.0018	0.0092	-0.0291
Elevation mirror	5001	-0.0012	0.0063	0.0010
Azimuthal mirror	5502	0.0014	0.0063	0.0877
Focusing mirror	5004	0.0140	0.0071	-0.0063
Secondary mirror	9001	0.0043	-0.0558	-0.0117

[1] These values were obtained by a rigid body rotation about the X axis of 3.13E-5 radians at the base of the alidade, a rigid body translation of 0.110 inches in the Z direction, and a rigid body translation of 0.004 inches in the -X direction.

VIII. Thermal Distortion Analysis

No thermal analysis has been performed to predict temperatures for the primary support truss. It is felt that the temperature environment will be severe caused by the laser beam energy which is absorbed into the cluster panels by 1) direct impingement due to the gaps (1-2% of the total area) between the individual segments, and 2) conduction through the mirror assemblies due to

absorption of the mirror coating. With the advent of the cluster panel metrology system, it will be possible to accommodate distortions normal to the surface caused by thermal gradients or bulk temperature changes. However, radial distortions can not be accommodated by this cluster panel control system. For the edge sensors to work, the gap between individual mirror segments must be maintained within certain tolerances. Preliminary edge sensor work indicated an allowable motion of only ± 5 microns (0.0001968 inches). If the mirror segments

on different cluster panels must maintain their spacing, **then** the primary mirror support truss must limit the radial distortions to small displacements. By using a graphite-epoxy tube and **steel** end fitting system, the **CTE** of the **tube/fittings** assembly could be lowered from 13.0×10^{-6} in/in/°F for aluminum to 1.5×10^{-6} in/in/°F. This system would allow a temperature gradient of 3.29°F between adjacent panels. The initial installation of the 150,000+ segments would have to account for the radial motion caused by the bulk temperature change from room temperature to the nominal operating temperature (122°F for example). Fans or other thermal control systems would need to be employed to help minimize temperature gradients throughout the primary mirror support truss.¹⁵

IX. Other considerations

Design and analysis considerations must be given to additional topics that were not sufficiently **analyzed** in the initial phase of this project. Topics regarding dynamic interaction, momentum compensation of optic elements, water-cooled optical **elements** and their associated jitter, transparent solid windows in domes, and fabrication considerations are briefly mentioned below.

Dynamic Interaction Concerns

In order for the control system to operate efficiently, the frequencies of the actuator commands must avoid coupling with the frequencies of the cluster panels and primary mirror support truss.

interactions between the actuators and the cluster panels occurs at two levels. The first **level** is at the individual segment location where interplay between the actuator forces and the cluster stiffness can lead to stroke inefficiency. Stroke efficiency of the actuators will be a **big** issue due to the limited stroke of the actuators. As the mirror is actuated up and down, forces in the actuator will be reacted into the cluster panels. For maximum **efficiency**, the effective mass of the panel at high frequencies must be much greater than the effective mass of the mirror segment. An estimate of the effective mass can be made with the following **formula**.¹⁶

$$Mass_{effective} = \frac{8\rho_s K C_L}{2\pi f}$$

wherein: ρ_s = mass per unit area (lbs/ft²)

K = radius of gyration (feet)

c_L = wave speed = $(E/\rho)^{1/2}$ = 16831 ft/sec
(for aluminum or steel)

f = operating frequency of actuator (hz)

Preliminary calculations have shown a 93% stroke efficiency for the baseline design.

The second level of interplay deals with resonant modes of the cluster panel itself. Passive damping augmentation should be used to effectively reduce, the dynamic interaction between the panel and the mirror segments. A simple and effective form of passive damping exists with constrained layer damping wherein a special backing material layer is bonded to the back surface of the support panel. When loading normal to the surface occurs (see Figure 6), shear loads are transmitted through the

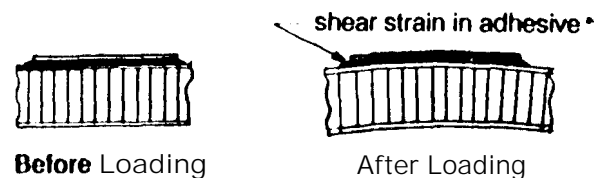


Figure 6. Example of constrained layer damping

adhesive material into the backing layer thereby exercising the specially formulated adhesive material. The shear strain energy is transformed into heat energy through viscous dissipation, thereby providing damping.

Interactions between the mirror segment motion and the entire primary **mirror** support truss are assumed to be negligible. A proposal to validate the above statement by testing was presented in 1992. Figure 7 shows the test configuration. One of the scheduled tests would be to **quantify** the interaction between the primary mirror truss and individual mirrors or groups of mirrors when subjected to a variety of **controlled** dynamic motions of the mirror segments. For example, a standing wave which

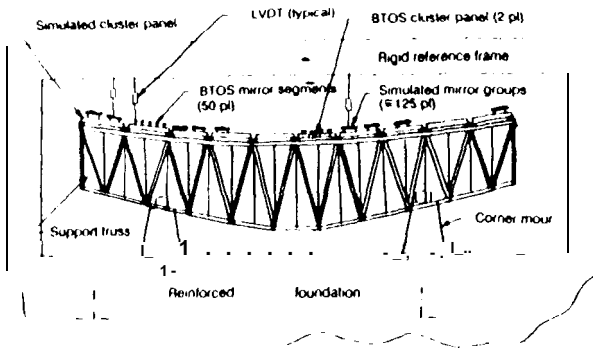


Figure 7. BTOS dynamic interaction test configuration

represents a typical atmospheric disturbance (gust of wind) could be commanded at various frequencies. The resulting motion of the primary mirror support truss could then be measured to determine the interaction effects. The proposal would also validate that manufacturing of the support was feasible.

Momentum Compensation of Optical Elements

A high speed tip/tilt mirror will be used to correct for rigid body rotations associated with atmospheric disturbances. Because the mass of the mirror (current baseline is to use the elevation mirror) is over 50 kg, it is felt that a momentum compensation device will be necessary to avoid dynamic interaction with the primary support truss. The frequency of operation could be as high as 300 cycles per second. Experience has shown that this problem is not trivial,

Water-cooled Optical Elements

Due to the extreme heat energy involved with the laser beam and mirror coatings with some absorptivity, active cooling of certain optical elements will be necessary. An optical mirror which operates at temperatures much higher than the surrounding atmosphere will generate significant thermal blooming effects which will adversely effect the demands on the mirror segment actuators. Active cooling of the azimuthal mirror, elevation mirror,

and secondary mirror will be required to help minimize the temperature differences. One method involves flowing cool water through channels in the back side of the mirror to help transfer the heat away from the mirror. The choice of mirror material must consider stiffness, thermal conductivity, and thermal distortion.¹⁷ If the flow of the water causes turbulence in the channels, then small vibrations (jitter) can occur. These effects must either be compensated by the individual mirror segment actuators or eliminated by proper hydraulic design,

Transparent Solid Windows

One of the initial assumptions for the BTOS design, required a nitrogen atmosphere be present in the volume between the primary and secondary mirrors. Initial designs assumed an enclosed "tube" around the perimeter of the primary mirror extending up to the secondary mirror. If this design were left unprotected by side winds, the resulting motions of the tipping structure would place heavy demands on the cluster panel metrology system and the individual mirror segments.

To avoid overloading the tipping structure, one idea called for a solid window to be placed in a moving dome structure. This concept would allow the entire dome structure to be purged with dry nitrogen and would keep all wind loads from affecting the tipping structure. The dome would hold this solid window and would track the motion of the tipping structure. A conventional thick, transparent, non-absorbing window (7" fused silica) could weigh over 84 tons. A lightweight film stretched across the aperture could possibly work. Some materials for the composition of this thin window have been suggested.¹⁸ Problems associated with any 14 meter diameter window need to be studied further.

Manufacturing Concerns

Unrelated to structural considerations is the problem of assembling 150000 + mirror segments to tolerances less than .001" relative to each other. It is felt that a robotic tool could be made cost effective if its design not only determined the correct position of each mirror assembly, but also bonded the assembly to the cluster panels. By starting at the

center of the primary mirror and working radially outwards, the job of bonding these mirrors in place could take less than one year! Each of the 156 cluster panels could later be removed one at a time (with approximately 1000 **mirror** segments each panel) for refurbishment or calibration.

X. Conclusions

The conceptual phase study for the Beam Transmission Optical System has been concluded. Structural design and analysis has led to a baseline structure design which meets its known requirements. A great deal of effort needs to be devoted to quantifying the dynamic interaction between the individual mirror segments, cluster panels, and primary mirror support truss. Thermal distortions caused by temperature gradients could overwhelm the distortions caused by gravity, unless careful thermal management and proper selection of structural materials are imposed.

Structural challenges also exist in areas such as the jitter associated with water-cooled optics, and momentum compensation devices associated with high speed tip/tilt mirrors,

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